Passive Thrust Oscillation Mitigation for the CEV Crew Pallet System

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Abstract

The Crew Exploration Vehicle (CEV) was intended to be the next-generation human spacecraft for the Constellation Program. The CEV Isolator Strut mechanism was designed to mitigate loads imparted to the CEV crew caused by the Thrust Oscillation (TO) phenomenon of the proposed Ares I Launch Vehicle (LV). The Isolator Strut was also designed to be compatible with Launch Abort (LA) contingencies and landing scenarios. Prototype struts were designed, built, and tested in component, sub-system, and system-level testing. The design of the strut, the results of the tests, and the conclusions and lessons learned from the program will be explored in this paper.

Introduction

The Constellation Program aimed to send human explorers back to the Moon and beyond as part of the Vision for Space Exploration. The CEV, also called Orion, was the proposed human spacecraft capsule for the program. The launch environment of the proposed mission included the TO phenomenon described below. It had been determined that the effects of the TO event must be mitigated for crew safety and operational reasons.

<u>CEV Background</u>

Originally conceptualized as a six-man vehicle, the CEV was ultimately designed to support a four-man crew on trips to the Moon, Mars, and other destinations in the solar system as part of the Constellation Program. The interior of the Crew Module (CM) portion of the CEV contains a floating crew pallet structure supported by eight struts upon which the crew seats are installed. See Figure 1 for an image of CM interior design and crew seat pallet with struts. The goal of the NASA Engineering and Safety Center (NESC) team was to develop a Crew Impact Attenuation System (CIAS) to mitigate the TO effects on the crew.

<u>TO Description</u>

Like all launch vehicles, the launch vehicle dynamic environment is a significant input to the overall payload launch loads. The Ares I First Stage has a small oscillation in thrust at a frequency band centered at approximately 12 Hz +/- 2.5 Hz, for about 10 seconds late in booster burn. Since the overall launch vehicle resonated with the input frequency of the boosters, the crew pallet struts had an additional requirement to mitigate the thrust oscillation resulting from the vehicle resonance.

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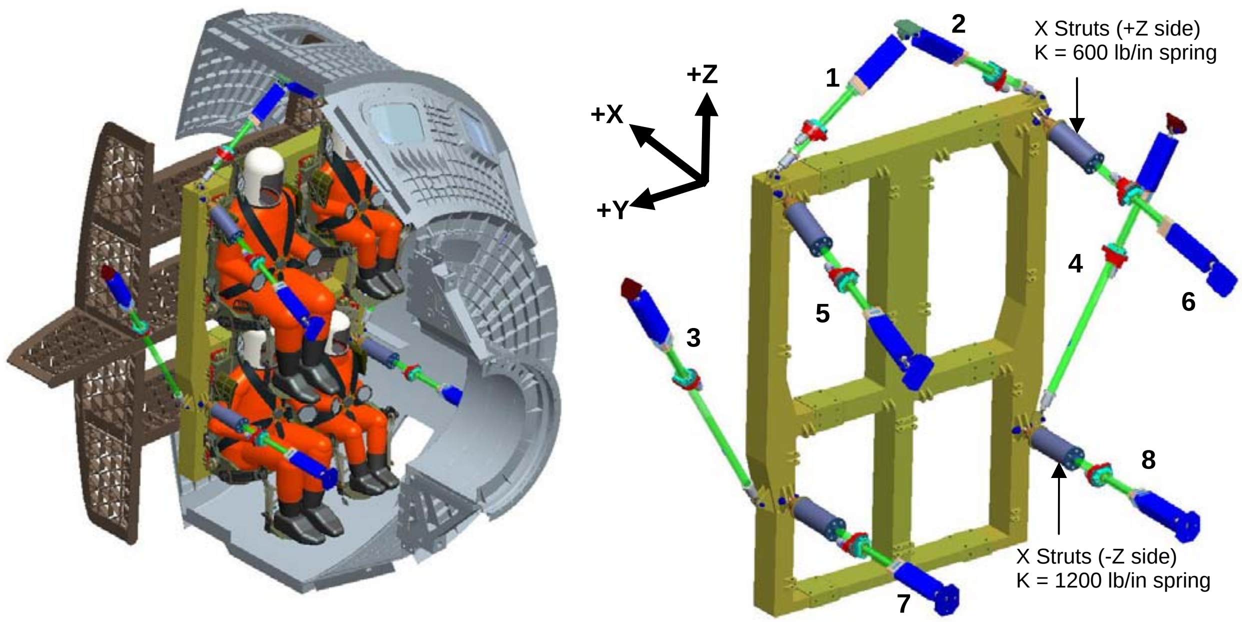

Figure 1. CEV CM with Crew (left); Crew Pallet with Struts (right)

Requirements

The TO isolation solution was driven by the requirements and constraints imposed on it derived from evaluations of a spring-based isolation system's impact on the crew in a TO event, LA scenario, and landing scenario. In addition, volumetric constraints and project constraints (resources, time) led to the design solution. This section will detail the results of the TO event, landing scenario, and LA evaluations.

CM Interior Environment

There are eight struts total: four in the X direction, launch axis, and two each in Z and Y directions. The CM baseline strut was modeled and tested using wire bender struts that absorb landing loads by dissipating energy through plastic deformation of steel wires – they have a one-time stroke during landing. As a pallet-based isolation system was selected (as opposed to seat-based), it was determined that the struts would need to be in series with the wire bender struts. Analyses showed that TO Isolators only needed to be present in the four X struts.

Crew Impact from TO Event

A requirement resulting from an investigation by the TO Focus Team and imposed by the Crew Office was to reduce crew response acceleration levels during the TO event to 0.25 g maximum at 12 Hz. This became the primary design driver for the isolation system.

The NESC team used the Brinkley Dynamic Response model to assess the likelihood of injury to the crew in both the landing and LA scenarios. For more information regarding the Brinkley injury risk criteria, see Reference 1.

Isolation Frequency

Knowing the TO frequency band of 12 +/- 2.5 Hz, a NASTRAN® coupled loads model was used to evaluate the effectiveness in g-reduction at different isolation frequencies. Following the theory of load transmissibility, if the pallet system frequency is greater than that of the input, dynamic amplification will occur. By dropping the pallet system frequency below that of the input, dynamic amplification can be eliminated and transmissibility can be less than 1.0.

It was concluded that an isolation frequency of 5 Hz or less is required to reduce the crew acceleration responses to 0.25 g. A 4.5 Hz isolation frequency was selected as optimal, as it represents a good

balance between deflection and transmitted dynamic load. This isolation frequency of the pallet is achieved by placing linear springs in series with the 4 X-axis struts.

Strut Deflection

During the TO event and while isolated at 4.5Hz, the pallet will translate between 0.508 cm – 1.016 cm (0.2 in – 0.4 in). Therefore, the TO Isolator stroke was designed to allow for slightly more than the predicted max of 1.016 cm before hitting a hard stop. In order to eliminate Isolator deflection during ground operations, the mechanism was designed to be preloaded through launch until the TO event occurs between 3.25 g's to 4.5 g's quasi-static load.

Volumetric Constraints

Volumetric constraints were imposed by the available strut length and the crew size. It was determined that, being in-line with the CM struts, a maximum length of 39.37 cm (15.5 in) was not used for the landing event and allowable for the TO Isolator design. The cylindrical diametric size had to be kept to a minimum to ensure that it did not contact a crew member's shoulder. Computer Aided Design models proved this to be less than 11.43 cm (4.5 in).

Isolator Impact on Landing Loads

LS-DYNA®, kinematic analysis software, was used to investigate the effect from the TO Isolator on landing loads experienced by the crew and compared to the baseline.

The initial model consisted of CM struts in series with an isolation spring; subsequent models included a damper to address potential issues with LA. With a 30g input, it was found that introduction of the TO isolation springs does not significantly or detrimentally affect accelerations transmitted to the pallet. However, the isolation springs have a detrimental effect on the stroking of the baseline struts, which ideally should be held to a minimum. Preliminary analyses showed that the Isolators need to be locked out during a landing event to minimize CM strut stroke while keeping the Brinkley model's injury risk probability to an acceptable level.

Isolator Impact on LA Loads

Although other launch analyses were done using NASTRAN®, the effect of the Isolators on crew loads in the contingency case of a LA were evaluated with LS-DYNA® software.

This was done so that crew response of the abort event could be determined while keeping the Isolator kinematic motion in the model. Without damping, the seat accelerations were found to be unstable and grew without bounds. With just 44.5 N (10 lbf) of Coulomb friction in parallel with the Isolator, the accelerations were sufficiently attenuated and decayed once the abort loadings ended. Since actual damping levels throughout the crew module are unknown and difficult to determine it was not useful to attempt a detailed damping study for the isolation springs. Instead, it was decided to utilize a small amount of Coulomb damping in the model and then insure that the design of the isolation spring had a mechanism for providing a deterministic level of damping.

TO Isolator Strut Design

Design Overview

The TO Isolator strut was designed as a passive spring and damper system that would be active during ascent and locked out during landing. It is mounted in-line with the X-axis CM struts. For hardware testing, the Isolator was mounted to the wire bender via a 1.905 cm – 16 (750"-16) threaded interface, see Figure 2.

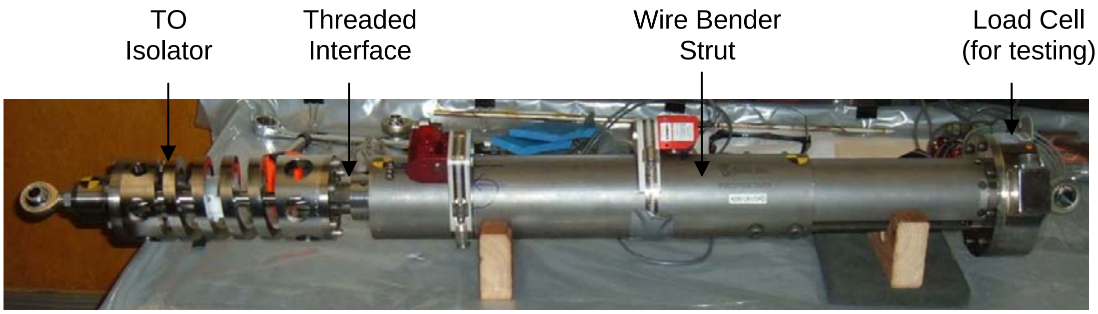

Figure 2. TO Isolator Strut In-line with Wire Bender Strut

The principal details of the TO isolation design are:

- Two 105.076 kN/m (600 lbf/in) springs (+Z) and two 210.151 kN/m (1200 lbf/in) springs (-Z) were required to achieve a balanced 4.5 Hz pallet system frequency.
- Springs are extended 4.123 cm (1.625 in), preloading the Center Rod in compression, so that the Isolator does not unseat prior to the TO event. During launch, the rod will not unseat from the housing thereby keeping the springs out of the load path until just before the TO event's quasi-static load is reached. The pallet then floats at 4.5 Hz for the range of 3.25 to 4.5 g's.
- During the TO event, cyclical motion of the Isolator will occur in the range of 0.508 cm – 1.016 cm (0.2 in – 0.4 in). The design allows for motion up to 1.588 cm (0.625 in).
- In the preloaded condition, the Isolator design does not affect the baseline strut stiffness.

<u>Design Detail</u>

The overall Isolator architecture is a large machined spring housed between two hubs. Threaded into and protruding from the exterior of the Front Hub is a threaded rod end with a spherical ball joint at its end. The interior of the Front Hub has a large counter-bore upon which a Delrin® pad sits, providing a contact surface for the Center Rod, which is preloaded against the pad. The Center Rod passes through the center of the spring and screws into the Back Hub which also provides a threaded interface to the wire bender strut. The concentric alignment of the entire Isolator assembly is controlled by a Flanged Sleeve which is pinned to the Front Hub at assembly and bolted between the front hub and the machined spring. Inside the flanged sleeve are some tight tolerance Delrin® tube bushings which provide a slip fit guide for the Center Rod. Refer to Figure 3 for a cross-section view of the Isolator assembly.

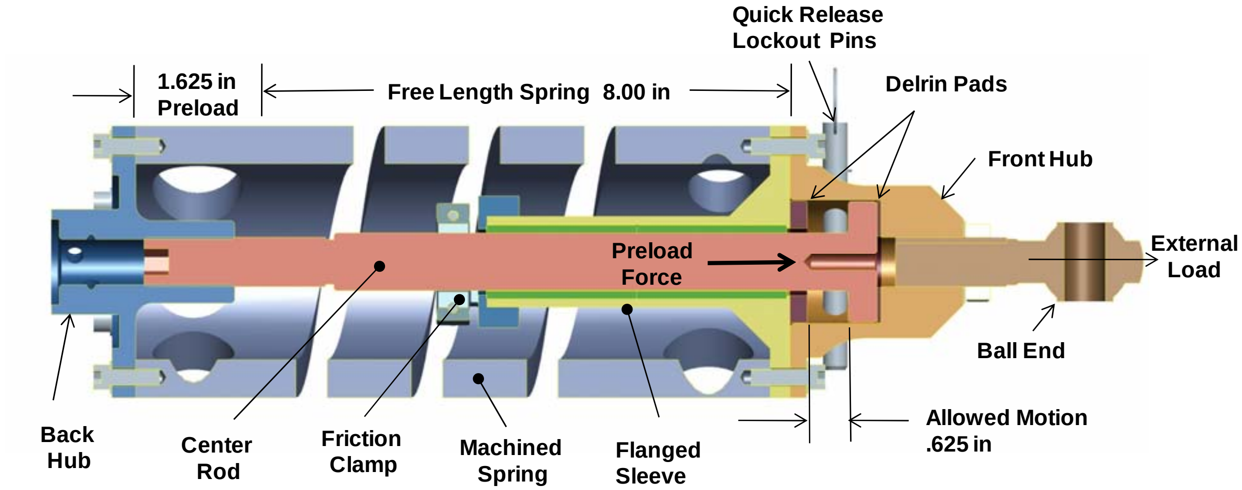

Figure 3. TO Isolator Strut Cross-Sectional View

During assembly, each spring is pulled back 4.123 cm (1.625 in) from its free length. The reactive force of this tensile load on the spring compresses the Center Rod onto the front Delrin® pad and preloads the assembly. During launch, when a large enough tension load is applied to overcome the preload, the center rod will unseat. After unseating as the spring extends further under still higher tensile launch loads, the relative motion of the Center Rod may travel 1.588 cm (0.625 in) until it hits another Delrin® pad stop on the Flanged Sleeve. The Delrin® pads were used so that titanium-on-titanium contact would not occur during impacts, preventing galling of the contacting parts. The expanded diameter head on the front end of the Center Rod limits its motion to this 1.588 cm (0.625 in) travel space. The Ares first stage forcing function has been computed to apply tensile loads resulting in head travel within this 1.588 cm (0.625 in) range, effectively isolating the pallet from the rest of the crew module in the X direction.

All of the load bearing machined parts and the machined spring are made out of titanium to reduce the weight of the assemblies. Each of the machined springs were machined from a single piece of titanium and have identical bolted interfaces on both ends. Due to the method of construction and the bolted interfaces the springs can react both compressive and tensile loads. The 105.076 kN/m (600 lbf/in) Isolator spring assemblies each have a mass of 5.08 kg (11.2 lbm) and the 210.151 kN/m (1200 lbf/in) Isolator spring assemblies each have a mass of 6.21 kg (13.7 lbm). The total mass for all four Isolator spring assemblies is 22.59 kg (49.8 lbm) which does not include the existing CM portion of each strut.

<u>Damping</u>

To provide for the low-level damping required to maintain stability in a LA scenario, the motion of the Isolator requires a damping component. Due to the late addition into the design, a passive, friction-based damping system was devised. Two Delrin® components clamp on the Center Rod; fasteners squeeze the components together, providing a friction force against the Center Rod that can be controlled based on the fastener torque. They are designed to provide between 111.2 – 155.7 N (25 – 35 lbf) of frictional force.

<u>Lockout Mechanism</u>

In order to satisfy the locking requirement for landing loads, a feature was added to prevent any stroking of the Isolators. For ease of prototype testing and assembly, quick release (insertion) pins were inserted behind the head of the Center Rod preventing it from moving and creating a rigid load path that bypasses the machined spring.

For flight, a quick-reacting lock out mechanism would be required during landing. A concept for a Non-Explosive Actuator (NEA) driven blade was added to the design. The concept uses an NEA to hold a spring-loaded Plunger that is threaded into the NEA nut; when actuated by a 4 amp electrical signal, the NEA releases the Plunger which wedges behind the Center Rod, securing the Rod in place and creating a single shear load path that bypasses the spring creating a strut stiffness equal to that of the CM baseline strut. A second blade 180° from the first would be needed for redundancy. The concept used for testing incorporated an NEA mockup on one of the four Isolators to characterize its static performance. Refer to Figures 4 and 5 for images of the Isolator and NEA Lockout Mechanism.

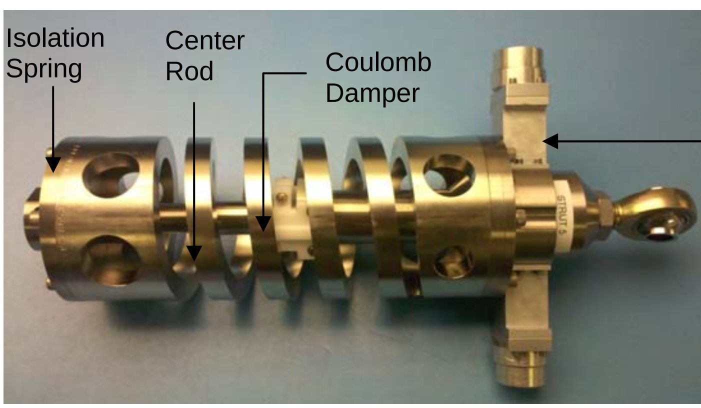

Figure 4. TO Isolator Strut with NEA Lockout Mechanism

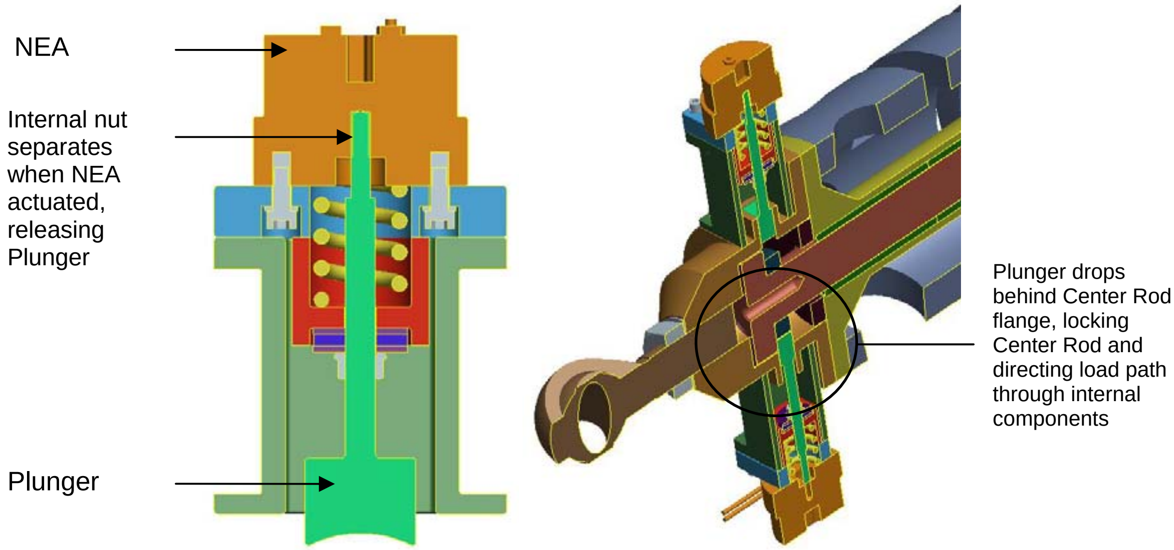

Figure 5. NEA Lockout Mechanism Cross-Sectional View

Test and Evaluation Program

The test and evaluation program aimed to characterize the effectiveness of the Isolation springs in attenuating the launch dynamic accelerations and loads imparted to the crew pallet. In addition, the test program aimed to understand the effect the Isolators had on the wire bender stroke during landing impact and better understand the correlation of model predictions to test results.

The test program included: Isolator Proof Testing, Isolator Stiffness and Damping Characteristics, Strut Impact Testing, System Drop Testing, Modal Testing of the launch configuration, and Vibration Testing to characterize performance.

<u>TO Isolator Proof Testing / Stiffness and Damping Characteristics</u>

Each Isolator assembly was tested in tension (locked and unlocked) and in compression to prove strength margins and verify spring stiffnesses. Maximum loading expected on the strut occurs during drop testing and is 4.448 kN (10,000 lbf). A test factor of 1.25 was included in proof tests leading to testing in compression and tension at 5.560 kN (12,500 lbf). All proof tests were performed successfully with no signs of failure or yielding. Spring stiffnesses and friction forces were found to be in acceptable ranges.

<u>Strut Sub-System Impact Testing</u>

Impact testing aimed to characterize TO vibration isolation springs in a dynamic environment and assess their effect on the behavior of the wire bender landing attenuation struts. In addition, an accurate LS-DYNA® model of the vibration isolation spring and attenuation strut was desired so that a reliable analytical model is available for subsequent landing simulations performed with the Orion vehicle under simulated landing conditions.

The tests were conducted using the 711th Human Performance Wing's Vertical Deceleration Tower (VDT) at Wright Patterson Air Force Base in Dayton, Ohio. The VDT consists of an 18.3 m (60 ft) vertical steel tower that allows a carriage to enter a free-fall state (guided by rails) from a pre-determined drop height. The plunger mounted on the rear of the carriage is guided into the hydraulic deceleration device (cylinder filled with water located at the base and between the vertical rails), producing an impact deceleration pulse, see Figure 6.

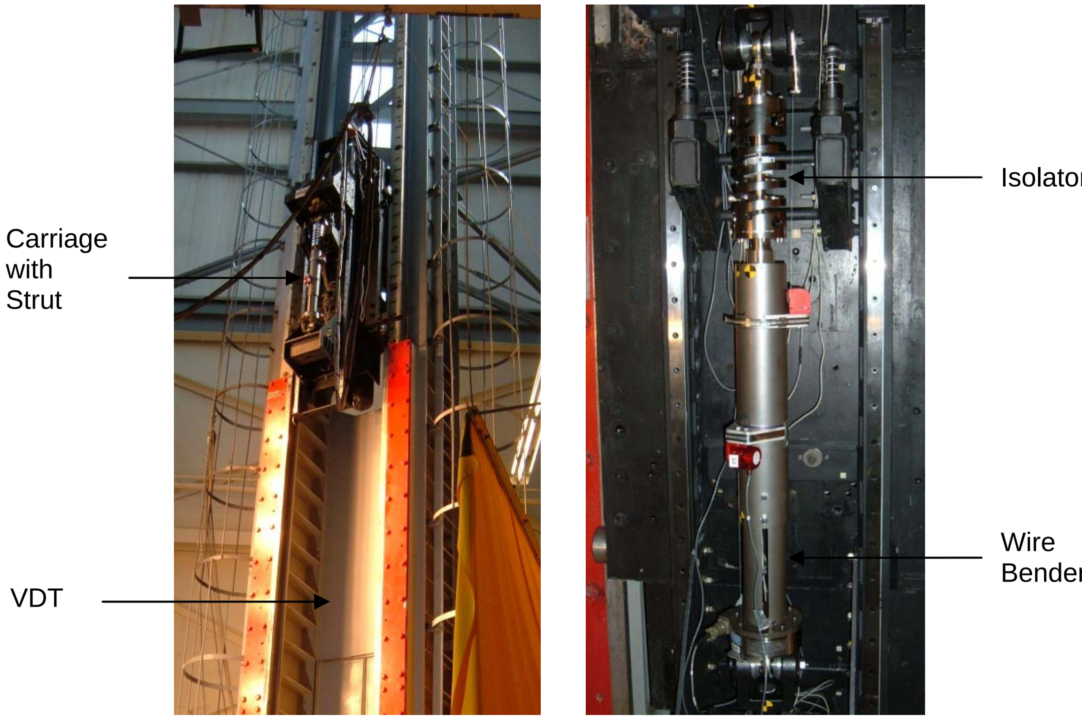

Figure 6. Attenuation Struts in VDT Setup

The testing showed that the transmitted acceleration peaks are similar regardless of the presence of the isolation spring. Testing and simulations for the strut with both the TO vibration isolation spring and the wire bender incorporated into the strut showed that the isolation spring has the effect of reducing the Brinkley injury criteria at the expense of increasing the strut stroke.

<u>System Drop Testing</u>

Orion CIAS impact tests were performed at the NASA Langley Landing and Impact Dynamic Research Facility to provide a demonstration of strut system performance, evaluate the performance differences between the locked and unlocked condition of the TO Isolators during the landing impacts and assess LS-DYNA® modeling techniques and predictive capability. The test configuration had a crew pallet-mockup (green, Figure 7) suspended from a cage (yellow, Figure 7) via the eight struts simulating the Orion CM interior. The cage was housed in a ring base/support structure to provide adjustment for various landing parachute pitch angles. Vertical drop tests were conducted by releasing the test fixture from a crane hook at a height calculated to produce the desired impact velocity. Tapered stacks of paper honeycomb at the four corners of the test fixture base were used to produce impact pulses approximating Orion water landings (see Figure 7). Measurements recorded during the tests include TO Isolator displacements, forces and accelerations as well as pallet and cage accelerations. High speed cameras and photogrammetry were used to verify impact conditions and observe TO behavior.

Eleven CIAS system drop tests were performed, which successfully demonstrated the performance of the system of struts and provided data for evaluation of the effect of the locked/unlocked condition. The accuracy of the LS-DYNA® model was also assessed. The tests featured impact velocities ranging from 3.05 - 10.67 m/s (10 - 35 ft/s) with the crew pallet locked at a 28^{o} pitch angle. The findings from this test and simulation effort are as follows:

1. Strut force and pallet acceleration time histories can be predicted via LS-DYNA® simulations with a high degree of accuracy and are relatively insensitive to expected variations in strut parameters such as strut load limit levels, initial stiffness, and dead zones (initial slack). The load limit in the struts determines the peak acceleration of the pallet. The expected range of the strut force limits will result in a relatively minor variation in the strut forces and pallet accelerations.

2. The strut stroke is the most important parameter to consider for evaluation of the system response. It is also the most difficult output to predict, due to its high sensitivity to most input variables. Comparisons of test data with the LS-DYNA® simulation results for tests 3 through 11 had an average prediction error in the strut displacements of ± 0.66 cm (0.26 in). The largest observed strut displacement error between a test and simulation was 3.81 cm (1.5 in). The overall average error was 20%. Accurate prediction of the strut strokes requires a high level of fidelity in the modeling of the structure to capture the flexural response of the crew pallet and the structure supporting the outboard ends of the struts, as well as very accurate modeling of the energy-absorbing wire bender strut force versus displacement curve.

3. Depending on the ratio of the load limit magnitude of the wire bender struts and the stiffness of the TO Isolator, there can be an amplification of the wire bender strut strokes for the unlocked condition. The testing revealed that there are combinations of wire bender struts and Isolator struts where the unlocked condition of the Isolator struts does not result in amplification of the wire bender strokes, and other combinations that can amplify the wire bender strokes by a factor of 2.5 to 3.0. These results confirmed the LS-DYNA® predictions that the Isolator needed to be in a locked configuration during landing.

4. The LS-DYNA® model is accurate enough to be used as an effective design tool for further CIAS studies. The design uncertainty on the pallet acceleration environment will reflect the expected variation in the strut yield force and is expected to be limited to 10% provided that the struts do not exceed their stroke limits. For the strut strokes, a design margin of 20% should be used.

Figure 7. Attenuation Strut in Assembly (Left); Drop-Test Fixture (Right)

System Modal Testing

As part of the pre-test planning for the base-drive vibration testing of the CIAS isolation system, a series of modal tests were performed at NASA Langley to assess the pre-test finite element model (FEM) predictions. The objectives of the modal test were: (1) to investigate potential fixture modes in the frequency range of the vibration test (0-20 Hz); and (2) to verify FEM predicted modes for the isolated crew pallet.

Two Isolator test configurations were evaluated: (1) locked and (2) unlocked. In the locked configuration, the isolated struts perform like rigid elements. For the unlocked configuration, the Isolators were shimmed to position with the Isolators in their active stroke range. Accelerometers measured the Frequency Response Functions (FRFs), calculated as the ratio of the acceleration response to the input force. Modal parameters (natural frequencies, damping factors, and mode shapes) were then estimated from the FRFs. Base-drive data acquired during the vibration testing was also used to obtain modal estimates with and without the friction dampers, see Figure 8.

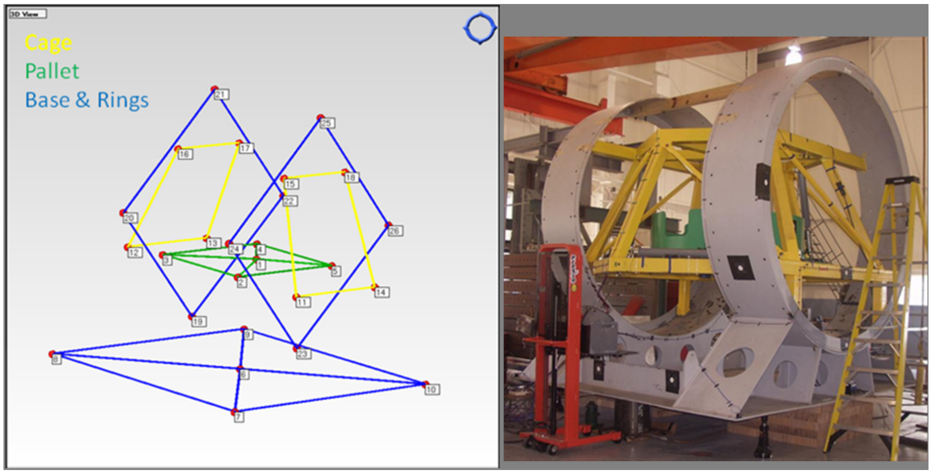

Figure 8. LS-DYNA® Model (Left); System-Level CM Mockup (Right)

Due to low-level inputs, at certain times it was difficult to overcome the friction in the damper, which created difficulties in test execution and data collection. Eventually the friction dampers were removed, allowing for more ideal performance of the Isolators; as a result all three pallet modes were identified. There is good agreement between the measured frequency estimates and FEM predictions with the largest frequency difference at 8% for the first mode. Modal results are presented in Table 1.

Table 1. Modal Estimates for the CIAS Pallet Modes

Mode	Predicted (Hz)	Modal with friction dampers		Base-Drive with friction dampers		Base-Drive without friction dampers	
		Freq (Hz)	Damp (%)	Freq (Hz)	Damp (%)	Freq (Hz)	Damp (%)
Rocking-Y:+Z struts	3.77	4.1	8.8	-	-	4.1	3.2
Twist about X	4.37	-	-	4.5	7.9	4.7	4.1
Rocking-Y: -Z struts	5.35	5.3	6.0	5.3	7.1	5.4	3.5

System Vibration

The objective of the system-level vibration test was to demonstrate the effectiveness of the pallet-to-strut isolation springs for reducing crew loads during a TO event occurring at approximately 12 Hz. The Orion wire bender Vibration Testing Unit (VTU) was subjected to dwell tests at frequencies of 10 and 12 Hz with varying amplitudes to represent a range of possible launch oscillation loads. Testing occurred at the Naval Surface Warfare Center (NSWC) Dahlgren Division Vibration Test Facility. See Figure 9 for a picture of the test setup.

The test consisted of three configurations:

- Test Sequence 1: rigid, Isolators locked out
- Test Sequence 2: Isolators released, friction dampers installed
- Test Sequence 3: Isolators released, friction dampers removed

Each configuration was subjected to three sine sweeps at varying input levels to test linearity. After the sine sweeps, each configuration underwent dwell testing at the TO frequencies with three different input levels: 0.2, 0.35, 0.5g.

Figure 9. Orion VTU Mounted on Hydraulic Shakers in the NSWC Vibration Facility

Conducting the signature sine sweeps verified the linearity of the system for Test Sequences 1 and 2. Because dampers were removed for Test Sequence 3, acceleration limits were reached as the pallet isolation frequency coupled with the input. As a result, only one successful sine sweep at 0.025g was completed over the range 2 – 8 Hz for Test Sequence 3. The test was modified to only sweep across 8 – 20Hz for levels higher than 0.025g (0.075 and 0.1g) which verified the linearity of the system over the range 8 – 20 Hz. However, the linearity of Test Sequence 3 over the range 2-8 Hz cannot be verified. Any slight deviation from linearity in Test Sequence 1, Test Sequence 2, and Test Sequence 3 (8-20 Hz only) can be attributed to the noise introduced by the hydraulic shakers.

Figures 10 and 11 display the reduction in accelerations attributed to the Isolators at the 10 Hz dwell and 12 Hz dwell. Tables 2 and 3 display the reduction in acceleration for the 10 Hz and 12 Hz dwells (note that a negative reduction indicates an increase in acceleration).

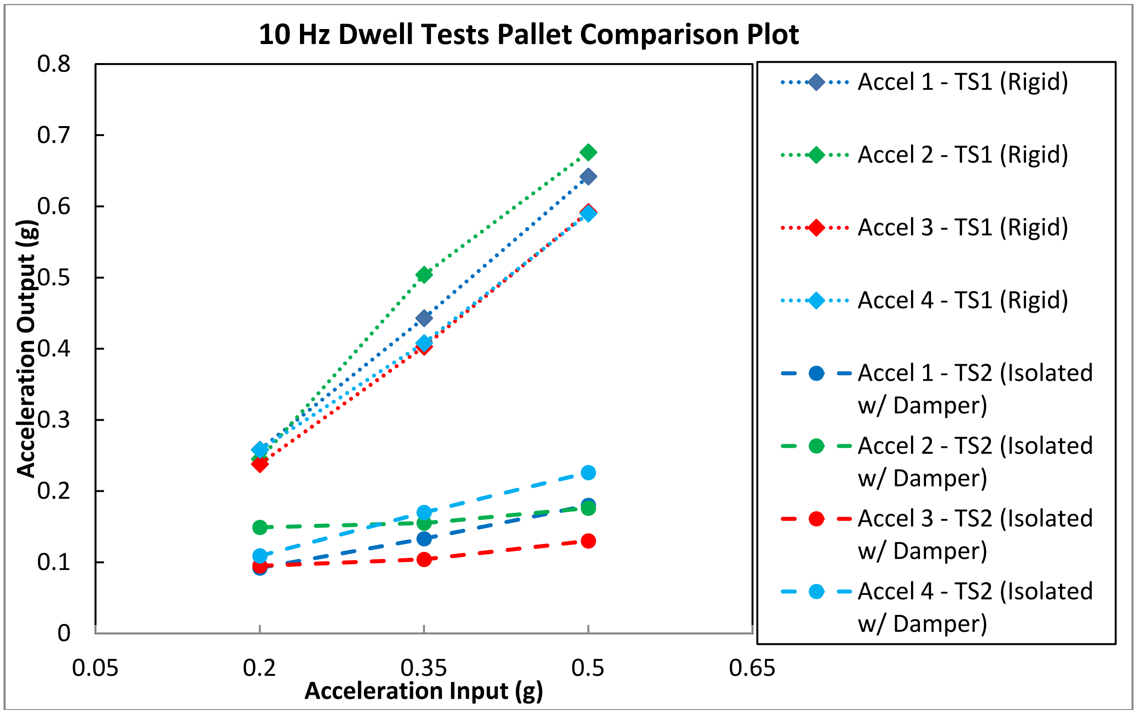

Figure 10. Comparison of Pallet 10 Hz Dwell Test Results for Each Test Configuration

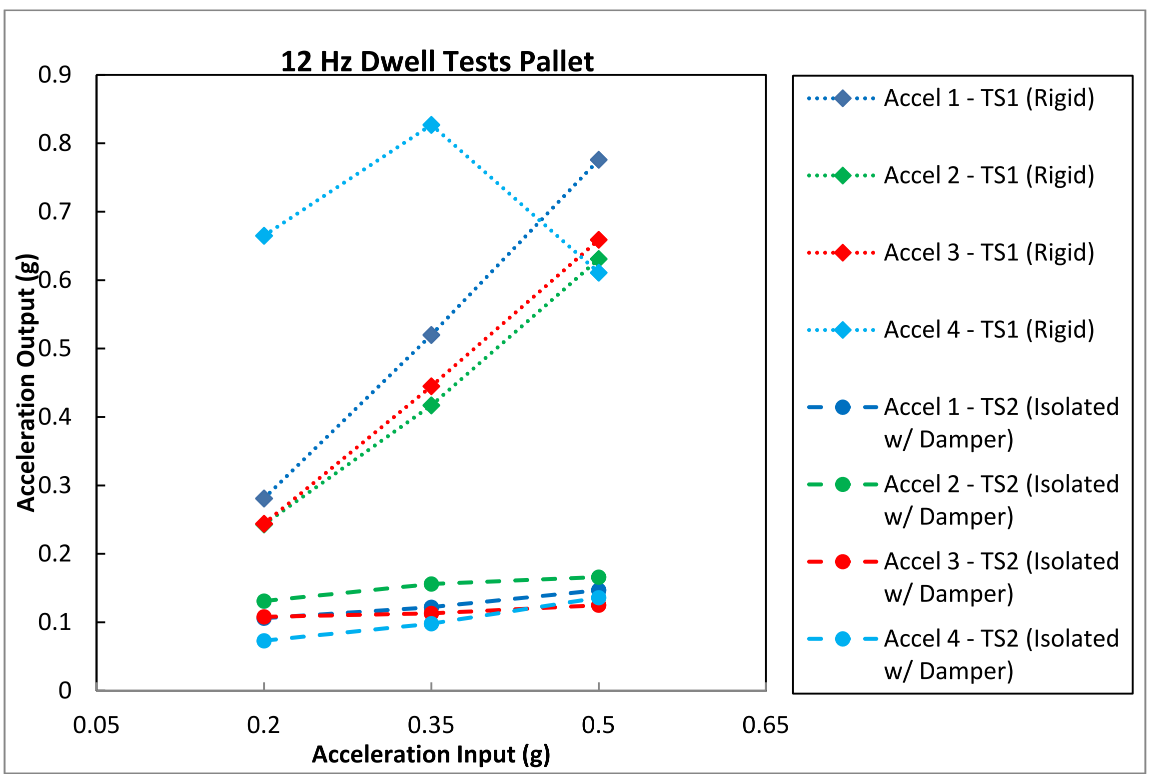

Figure 11. Comparison of Pallet 12 Hz Dwell Test Results for Each Test Configuration

Table 2. Average Pallet Accelerations for 10 Hz Dwell Tests

G Input	Test Sequence	Input as Measured (g)	Average Pallet Response (g)	Average Pallet Transmissibility	Average Pallet % Reduction Compared to Input*	Average Pallet % Reduction Compared to TS1 (Rigid)
0.2	TS1	0.203	0.258	1.27	-27.09	**N/A**
	TS2	0.2	0.092	0.46	54.00	**64.34**
	TS3	0.198	0.06	0.30	69.70	**76.74**
0.35	TS1	0.348	0.443	1.27	-27.30	**N/A**
	TS2	0.347	0.133	0.38	61.67	**69.98**
	TS3	0.355	0.112	0.32	68.45	**74.72**
0.5	TS1	0.501	0.642	1.28	-28.14	**N/A**
	TS2	0.506	0.18	0.36	64.43	**71.96**
	TS3	0.497	0.154	0.31	69.01	**76.01**

*Note: Negative reduction values indicate increase in input.

Table 3. Average Pallet Accelerations for 12 Hz Dwell Tests

G input	Test Sequence	Input as Measured (g)	Average Pallet Response (g)	Average Pallet Transmissibility	Average Pallet % Reduction Compared to Input*	Average Pallet % Reduction Compared to TS1 (Rigid)
0.2	TS1	0.198	0.281	1.42	-41.92	**N/A**
	TS2	0.197	0.106	0.54	46.19	**62.28**
	TS3	0.199	0.042	0.21	78.89	**85.05**
0.35	TS1	0.347	0.52	1.50	-49.86	**N/A**
	TS2	0.35	0.122	0.35	65.14	**76.54**
	TS3	0.344	0.073	0.21	78.78	**85.96**
0.5	TS1	0.507	0.776	1.53	-53.06	**N/A**
	TS2	0.5	0.147	0.29	70.60	**81.06**
	TS3	0.492	0.105	0.21	78.66	**86.47**

*Note: Negative reduction values indicate increase in input.

During Test Sequence 3, the average reduction in acceleration or loading remains relatively constant over all the input levels tests. Test Sequence 2 on the other hand seems to experience a greater percent reduction as the input level is increased. After the completion of Test Sequence 2, the Isolators were removed and the friction damper force was tested. The friction dampers for Isolators 7 and 8 were within the 111.2 – 155.7 N (25 – 35 lbf) target range while struts 5 and 6 had static/kinetic friction values of 266.9 N (60 lbf) and 66.7 N (15 lbf), respectively. During low level dwell and sine sweeps of Test Sequence 2, strut 5 was not fully released which resulted in a shift in the system dynamics.

The test demonstrated that the TO Isolators do an effective job at mitigating loads due to a TO event, reducing pallet accelerations to 20-40% of that of the input. The Delrin® friction dampers however, are overly sensitive and altered the dynamics of the system making correlation of pre-test analyses with the dampers difficult. This prototype damping system should be replaced with a more reproducible, controllable damper for a flight system.

Lessons Learned

Passive Friction Damping

The Isolator design solution utilized a passive, friction-based damping scheme. The inclusion of damping into the Isolator was not introduced until after fabrication of the Isolator parts, which partially drove the solution. Friction, by nature, is difficult to control and get repeatable results. The end result of this is that the friction force applied to the Center Rod, and therefore the damping coefficient of the Isolator, changes slightly from test to test. In addition, the Center Rod was turned down on a lathe, leaving a concentricity and roundness tolerance greater than desired. This caused variations in friction clamping force throughout Center Rod travel, which was observed in Isolator friction testing. A more appropriate finish on the Center Rod for a passive friction damping system would be to centerless grind the shaft, which would result in less surface variations and more consistent friction forces.

Structural Adhesive under Impact Loads

At both the forward and aft hard stop of Center Rod travel, a Delrin® piece would contact the Center Rod to prevent like-material contact. Hysol® EA 9394 structural adhesive epoxy was used to adhere the Delrin® to the substrate metal. This was selected due to the fact that it would require minimal modification to the components, ease of assembly, low-profile installation, and common usage in space industry. The forward Delrin® pad was bonded to the inner counter-bore on the Front Hub. The aft Delrin® pad was bonded to the rear of Center Rod head.

The Delrin® pads did not dislodge or displace during proof testing. In the dynamic tests (such as Strut Impact Testing, System Drop Testing, and System Vibration) it was observed after certain test runs that the Delrin® would no longer be bonded to its substrate metal. The high impact loads would break the bond causing the Delrin® component to be loose within the cavity. Therefore it has been observed that structural adhesives are likely not appropriate for dynamic impact loading. If it were to be re-designed, the Delrin® pads would be hard mounted with small flat head screws slightly recessed below the impact surface.

Displacement Data Capture in Dynamic Environment

As a critical parameter used for performance and model correlation, the testing program required displacement measurements to be captured during an impact. String-potentiometers were installed in parallel to the struts to record the motion of the impact attenuation wire benders with very good performance. But when installed to measure isolator displacements during single strut/isolator impact tests, the string pots demonstrated greater error band than expected; often giving measurement readings that were much higher than theoretical maximums. From examining high speed camera video, the string pots appeared to have insufficient response during impact onset resulting in over deployment during initial extension and out-of-axis string movement during retraction. The exact cause of this is indeterminate; suspected causes are inertial affects within the device making it inadequate for isolator dynamics during impact or lack of stiffness in the string-pot mounting brackets. Linear Variable Differential Transformers (LVDTs) using rigid plunger shafts were implemented for system drop and vibration tests and were found to perform well with reliable isolation deflection data.

Data Acquisition System Updating

Data from the Orion vibration test was captured using two data acquisition systems (DASs) with different capabilities. The Modal DAS was used to acquire the acceleration measurements on the test article. This DAS was specifically designed to capture and process data for near real-time viewing during the test. For the strut load cell and LVDT data acquisition an EME Corporation Model 3200L DAS was utilized. The EME DAS was designed to capture time data only and did not have any processing capability. The EME Corporation Model 3200L DAS could only be set to acquire data based on a fixed time interval. If a test ran longer than expected, the EME DAS time may expire requiring a re-test. As a result, the EME DAS was set to have a time approximately 5 minutes longer than the estimated testing time. The problem is, once the EME DAS starts, it cannot be stopped until the fixed time expires. This led to gaps between tests and acquiring data past the test's completion. It is recommended that an updated DAS be used for future tests, which do not rely on user defined data acquisition time frame, to save time between tests and to eliminate the acquisition of data past the test's completion.

Conclusions

The TO study confirmed the optimal crew isolation frequency of 4.5 Hz and testing established the system performance and damping mechanism value. From a load mitigation perspective, it was found that the pallet isolation approach was very appealing. Results indicated that the isolation system provided a reduction of dynamic load to about 20%-40% of the input and that Brinkley levels were met at a mass penalty of less than 5.897 kg (13 lbm) per strut.

The results of this test program illustrate the feasibility and benefits of implementing a pallet isolation system for the Orion CEV CM. The design and test data included herein are directly applicable to the Orion vehicle, but could be adapted to other designs with similar dynamic load issues. Isolation for load reduction is a flight proven technology utilized on several robotic spacecraft in addition to Space Transportation System payloads. Its low mass penalty is relatively insignificant when compared to the hardware benefits and the potential mass increases if this option is not exercised.

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